Department of Mechanical Engineering

Analytical analyses of active control of sound transmission through soft-core sandwich structures and double panel partitions

Kiran Chandra Sahu

Aalto University

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Analytical analyses of active control of sound transmission through soft-core sandwich structures and double panel partitions

Kiran Chandra Sahu

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Abstract

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Abstract

Active structural acoustic control (ASAC) is a form of active noise control which focuses on the control of structural vibrations in a manner that minimizes acoustic radiation from a structure. The greatest difficulty ASAC suffers from is in finding an "optimal" error quantity, which can be easily implemented in a control algorithm. Volume velocity control (VVC) metric which is generally used in ASAC typically requires either a large number of sensors distributed across the entire structure, or a single distributed shaped sensor. A new parameter termed "weighted sum of spatial gradients (WSSG)" showed a great potential to significantly reduce noise transmission using four sensors only. This thesis carried out a comprehensive study on VVC and WSSG, and numerical calculations indicate that, by intelligent selection of modal indices for the calculation of WSSG and the frequency band of active control, WSSG is able to achieve comparable amount of sound attenuation with VVC.

Soft-core sandwich panels are increasingly used because of their high strength-to-weight ratios. However, acoustical properties of these are less desirable at low frequencies, which can lead to high noise levels. Since additional sound absorbing materials fail to provide satisfactory results at low frequencies, active control techniques have been researched. It is very well known that soft-core sandwich panels vibrate in flexural modes (occur at low frequencies) and dilatational modes (occur in high frequencies). Therefore, in this thesis, VVC and WSSG control metrics are used to control these modes in order to achieve control in a broader frequency band. Numerical studies illustrate that the control metrics are able to attenuate both modes and increase the sound transmission loss irrespective of isotropic core loss factors.

In buildings, windows are often the weak link in protecting the interior from outside noise. In particular, double panel windows have a poor sound insulation at low frequencies around the mass-air-mass resonance. Sound absorbing materials in the air gap fail to provide satisfactory results in the low-frequency region, therefore, an active controller is an attractive approach to reduce the noise level in buildings. This thesis used the WSSG control metric to drive piezoelectric actuators in order to attenuate the dipole-type of motion of double panel systems, a motion that commonly occurs at high frequencies. Thereby, sound attenuation in a large frequency band can be accomplished. Also, acoustic point source arrangements are considered between the panels. Numerical results indicate that the actuator should be placed on the incident panel to minimize the control quantities on the radiating panel, and combined WSSG and cavity control is able to provide better sound attenuation in a wider frequency band.

Keywords Sandwich panel, double panel, soft-core, active control, volume velocity, WSSG, sound transmission loss

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Preface

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List of Publications

This thesis consists of an overview and of the following publications which are referred to in the text by their Roman numerals.

I. Kiran Chandra Sahu, Jukka Tuhkuri and J. N. Reddy. Active attenuation of sound transmission through a soft-core sandwich panel into an acoustic enclosure using volume velocity cancellation. *Proceedings of the Institution of Mechanical Engineers, Part C: Journal of Mechanical Engineering Science*, 229 (17), 3096-3112, 2015.

II. Kiran Chandra Sahu, Jukka Tuhkuri and J. N. Reddy. Active structural acoustic control of a soft-core sandwich panel using multiple piezoelectric actuators and Reddy's higher order theory. *Journal of Low Frequency Noise, Vibration and Active Control*, 34 (4), 385-412, 2015.

III. Kiran Chandra Sahu, Jukka Tuhkuri and J. N. Reddy. Active piezoelectric-structure acoustic control of a soft-core sandwich panel using volume velocity and a weighted sum of spatial gradient control metric. Accepted for publication in *Journal of Vibration and Control*, 2015.

IV. Kiran Chandra Sahu and Jukka Tuhkuri. Active control of sound transmission through a double panel partition using volume velocity and a weighted sum of spatial gradient control metrics. *Noise Control Engineering Journal*, 63 (4), 347-358, 2015.

Author’s Contribution

Publication I: “Active attenuation of sound transmission through a soft-core sandwich panel into an acoustic enclosure using volume velocity cancellation”

This paper presents an analytical study on active control of low-frequency sound transmission through a soft-core sandwich panel into a rectangular enclosure. Since noise transmission through a soft-core sandwich panel mainly occurs due to the flexural and the dilatational modes, a control strategy employing volume velocity cancellation is used to control these modes and achieve sound attenuation in a broad frequency range. The author carried out the analytical simulation, analyzed the results, and wrote the manuscript. Prof. Jukka Tuhkuri and Prof. J. N. Reddy contributed to the paper by giving valuable comments, suggestions and by assisting in the writing process.

Publication II: “Active structural acoustic control of a soft-core sandwich panel using multiple piezoelectric actuators and Reddy’s higher order theory”

This paper theoretically investigates the active control of low-frequency radiated sound power from a simply supported soft-core sandwich panel which is excited by a line moment. The governing equation of the sandwich panel is derived using the Hamilton’s principle considering Reddy’s third order shear deformation theory. The author carried out the analytical simulation, analyzed the results, and wrote the manuscript. Prof. Jukka Tuhkuri and Prof. J. N. Reddy contributed to the paper by giving valuable comments, suggestions and by assisting in the writing process.

Publication III: "Active piezoelectric-structure acoustic control of a soft-core sandwich panel using volume velocity and a weighted sum of spatial gradient control metric"

This paper presents an analytical study on active control of sound transmission through a simply supported soft-core sandwich panel, with the main focus on how weighted sum of spatial gradient control metric (WSSG) is able to achieve comparable sound reduction with volume velocity by intelligently selecting the modes in the calculation of scaling factors. The author carried out the analytical simulation, analyzed the results, and wrote the manuscript. Prof. Jukka Tuhkuri and Prof. J. N. Reddy contributed to the paper by giving valuable comments, suggestions and by assisting in the writing process.

Publication IV: "Active control of sound transmission through a double panel partition using volume velocity and a weighted sum of spatial gradient control metrics"

This paper presents an analytical study on active control of harmonic sound transmission through an acoustically baffled, rectangular, simply supported double panel partition. The effects of both structural and acoustic control were investigated. The author carried out the analytical simulation, analyzed the results, and wrote the manuscript. Prof. Jukka Tuhkuri contributed to the paper by giving valuable comments, suggestions and by assisting in the writing process.

1. Introduction

1.1 Background

In the field of acoustics, noise is generally defined as an unpleasant or disliked sound. This definition is straightforward, but the difference between sound and noise is by no means precise. For example, in the opinion of some (older) people, the sound of modern music is equivalent to noise. On the other hand, there are few who would say that the sound produced by passing traffic or a vacuum cleaner is pleasant.

Noise exposure of workers, particularly in industry, is one of the major health and safety problems which must be taken seriously. Long-term exposure at certain noise levels can lead to hearing impairment, hypertension, ischemic heart disease, annoyance, and sleep disturbance (Berglund et al., 1999). For example, a person who is exposed to noise exceeding 85 dB(A) on an average over a working day of 8 hours for a long period of time can suffer a permanent hearing loss (Fahy and Walker, 1998). Noise exposure can also create stress, increase workplace accident rates, and stimulate aggression and other anti-social behavior (Kryter, 1994). A recent WHO report illustrates that noise is the second major cause of human death in the Western Europe (WHO, 2011) even though the EU noise directives (European Union, 2002) have been enforced for quite a number of years. Due to these serious effects, the management and control of noise levels, especially in the workplace, is a subject of legislation.

The majority of industrial noise sources come from vibrating structures. Accurate prediction of sound radiation from such structures remains a challenging problem. Many structures can be presented in terms of an assemblage of flat plates. For example, machinery casings, car body shells, hulls of ships, walls, and floors. By reducing the complexity of such structures, that is, by approximating them to simple structures like plates, the mechanism of sound radiation can then be modelled considerably more easily with analytical or numerical approaches. The study of an isolated plate provides the basic understanding of the interaction between the vibration behavior of a structure and its sound radiation. From this, in many cases, the determination of sound radiation from more complex structures can be estimated reasonably accurately (Fahy and Gardonio, 2006).

Traditionally, noise was reduced by passive means like mufflers, noise barriers, damping plates, sound absorbing materials, double glazed windows, etc. The effectiveness of most of these passive means is limited to high frequencies, and all have some disadvantages. Mufflers in exhausts increase flow resistance and back pressure, which tends to reduce the machinery performance (Elliott et al., 1997). Damping plates and sound absorbing materials work best at high frequencies, but they are very large and heavy (Fuller and von Flotow, 1995). Due to resonances, low-frequency transmission loss of double-glazed windows can be worse than that of single-glazed windows (Cremer et al., 2005).

In the last two decades, active control of sound and vibration (at audio frequencies) has emerged as a viable technology to bridge this low-frequency technology gap (Fuller and von Flotow, 1995). The idea of active noise control was first conceived in the 1930s (Lueg, 1936). His idea is illustrated in Figure 1.1, which shows the control of a plane sound wave in a duct. Acoustic source "A" produces sound wave "S_1", which propagates through the duct (from left to right). The components of the active control system are microphone "M", electric controller "V" and loudspeaker "L". The signal measured by the detection microphone is passed through the controller to the loudspeaker. If the controller is tuned so that the speaker produces sound wave "S_2" with the amplitude of the original wave but shifted 180 degrees in phase, then the original wave is totally cancelled. A person at the right hand end of the duct would in that case detect no sound. An essential assumption for this to be true is that all components in the system behave in a linear way, that is, the principle of superposition applies.

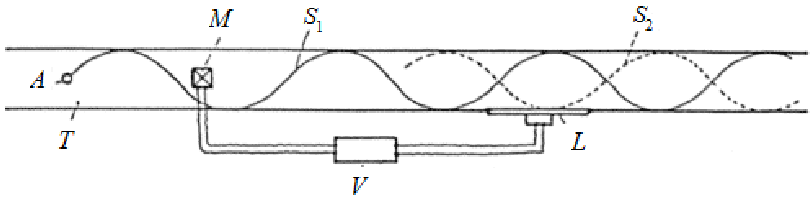

Figure 1.1. A figure from the illustration page of Lueg's patent (Lueg, 1936).

The concept of using a control system with speakers as actuators and microphones as sensors to reduce sound is referred to as active noise control (ANC). By means of destructive interference and impedance coupling the noise is reduced locally or globally (Nelson and Elliott, 1992). Practical use of ANC was limited for a long time because the technology was not available. For high levels of sound reduction, the amplitude and phase of the control signal must be accurate, which was difficult with analogue controllers. With the arrival of digital signal-processing techniques, ANC became more feasible. Nowadays, ANC technology is used for example in active headsets (e.g., by Sennheiser) and to reduce cabin noise in propeller aircrafts (e.g., by Lord Corporation).

A drawback of ANC is that when the acoustic source is distributed over multiple surfaces, as in vibrating plate-like structures, too many speakers are required to provide global control. Therefore, active structural acoustic control (ASAC) (Fuller, 1988; Fuller, 1990; Fuller et al., 1991) was introduced as an alternative to ANC. In ASAC, the control inputs are directly applied to the structure. The idea is to change the vibration of the structure with the objective of reducing the overall sound radiation. It is an extension of a technique called active vibration control (AVC) to the range of audio frequencies. Conventional AVC methods, which were developed parallel to ANC methods, were for instance used to control the vibrations of precision instruments in space structures. Such control problems are generally characterized by much lower frequencies than those used in noise control problems. ASAC also differs from AVC in that it attempts to control only the vibrations which are important to sound radiation (Fuller et al., 1996). In AVC, the vibration level is reduced to the best possible extent, with no concern regarding the overall sound. The benefits of ASAC over traditional sound-field control using a speaker arrangement are associated with the control of sound at the source and the system compactness. Structurally-applied actuators are much less intrusive than speakers because they do not use space in the acoustic field, which in some cases can be very valuable, as in confined cabins (Faber and Sommerfeldt, 2006).

The first references to ASAC concentrated on using point forces ("shakers") as secondary control forces and microphones as error sensors to control the sound radiated by a plate (Fuller, 1988; Fuller, 1990; Fuller et al., 1991a). More recently, distributed piezoelectric actuators have been used instead of point forces (Fuller et al., 1991b; Wang et al., 1991). A distributed piezoelectric actuator is a layer of piezoelectric material that is bonded to the surface of a structure. These actuators have the practical advantage that they can be integrated with the structure. As well as using actuators integrated with the structure, there has been growing interest in replacing error microphones by error sensors integrated within the structure (Maillard and Fuller, 1998; Sors and Elliott, 2002).

Although ASAC is potentially simpler than traditional ANC, it has inherited many of its problems. Two problems of particular significance for both techniques are the choice of sensor location (Nelson and Elliott, 1992) and the choice of the measurement quantity, or control metric, to be minimized. Without sufficient analysis, either of these could result in sound amplification. An important theoretical development that has guided research in resolving these problems is the radiation resistance matrix and its radiation mode shapes (Elliott and Johnson, 1993). The radiation resistance matrix was developed as an alternative to the Rayleigh integral for use in calculating the radiated sound power from distributed structures. Instead of integrating over a surface in the far field, as the Rayleigh integral does, the radiation resistance matrix method discretizes the surface of the structure into elementary radiators and then estimates the radiated power through the calculation of the self and mutual impedances between each elementary radiator on the surface (Fahy and Gardonio, 2006; Naghshineh and Koopmann, 1993). The radiation resistance matrix contains mutual radiation impedances, and its eigenvectors are the radiation mode shapes. The radiation mode shapes reveal the underlying sound radiation mechanisms of the structure. By targeting the radiation mode shapes of the structure, an error sensor on the structure can measure a quantity related to the radiated sound power of the structure and, therefore, is able to provide sound attenuation.

Several control metrics that target the radiation mode shapes of a distributed structure have been developed. At low frequencies, the first radiation mode shape, with a drum-like appearance, is the most efficient radiator. Volume velocity, which is calculated as the cross-sectional area of a space multiplied by the speed of the fluid flowing through it, is a measurable quantity on the surface of the plate and has a similar appearance to the first radiation mode (Elliott and Johnson, 1993). This has become the predominant control metric in the literature because of its success in providing sound attenuation (Johnson and Elliott, 1995; Gardonio et al., 2001; Sors and Elliott, 2002). Other research has developed a method that uses optimally-shaped piezoelectric sensors to target specific radiation or structural modes (Snyder et al., 1995; Snyder et al., 1996).

Some control metrics which do not target the radiation mode shapes have also been developed. For example, the most basic control metric is the measurement of squared velocity at a point on the plate (Sung and Jan, 1997). Energy-based control metrics have also been developed (Manwill et al., 2010). ASAC control metrics, which do not target radiation modes, have met with limited success at best.

Despite the variety of sensing techniques available, each of the methods described above has one of two common problems. Either they require a priori knowledge about the radiation or structural vibration modes of the system to be controlled, or they require a large number of sensors. For example, several methods have been developed to measure volume velocity. One method uses an array of accelerometers on the surface of the plate and estimates the overall volume velocity by summing the accelerometer outputs. Minimizing volume velocity is shown to achieve sound attenuation in certain instances, but several drawbacks exist. The primary drawback relates to the number of sensors required to accurately estimate the overall volume velocity of the vibrating structure. The total number of sensors required for a good estimation of the volume velocity is given by (Sors and Elliott, 2002),

$$N = \frac{5}{3\pi} cl \sqrt{\frac{m}{D}}, \tag{1.1}$$

where c is the speed of sound, l is the smallest plate dimension, D is the bending stiffness, and m is the mass per unit area. This means that, for a steel rectangular plate whose smallest dimension is 0.483 meters, the approximate number of sensors should be 62 (Fisher, 2010). This is impractical for experimental situations. Minimizing volume velocity is also shown to be ineffective for (even, even) structural modes. These structural modes have an equal amount of mass moving in the positive direction and in the negative direction, and so they have a net volume velocity of zero. These drawbacks limit the effectiveness of volume velocity as a minimization quantity. The increase in the number of sensors can be avoided by using a distributed piezoelectric sensor to measure volume velocity (Gardonio et al., 2001); however, this sensor would need to be designed for the specific geometry. Therefore, while the increased complexity from the sensor array is alleviated, the geometry dependence of the sensor requires a priori knowledge about the system. If, instead of volume velocity, the radiation mode shapes were targeted directly through the use of shaped piezoelectric sensors (Snyder et al., 1995), a priori knowledge would once again be required in order to shape the sensor properly for the given geometry. Finally, squared velocity, which requires only a single accelerometer to measure, is highly dependent on sensor location, and care must be taken not to place the sensor on a nodal line (Snyder and Tanaka, 1993); once again requiring a priori knowledge of the system. As a result of the difficulty in implementing control metrics in practice, ASAC has seen very little practical application.

From this brief survey of control metrics and measuring techniques, it can be concluded that in order for ASAC to live up to its potential as a simpler and more effective technique than traditional active noise control, a control metric which requires only a few sensors, is insensitive to sensor location, and does not depend on plate geometry, is necessary. A recent control metric, termed composite velocity (also referred to as weighted sum of spatial gradient or WSSG) has shown promise in resolving these issues. Composite velocity was developed as a weighted sum of spatial velocity gradients requiring only four sensors to measure. It was found to be relatively insensitive to sensor location on a simply supported plate, thus achieving the simplicity possible with ASAC and avoiding the requirement of priori knowledge (Fisher, 2010; Fisher et al., 2012). The effectiveness of this new control metric has also been experimentally realized in a flat simply supported plate (Hendricks et al., 2014). This motivates further research on WSSG and comparison of the results with volume velocity.

1.2 Objectives and scope

The objective of this thesis is to better understand the structural acoustic coupling, with the end goal of actively controlling structural vibrations in a manner that reduces the overall acoustic radiation. Therefore, active structural acoustic control procedure called "weighted sum of spatial gradients (WSSG)" control metric is used, the results are compared with the “volume velocity” and the relation between the two control metrics is also investigated. In this thesis, only simply supported boundary condition and isotropic materials are considered in order to compare the results with the already published articles in the literature. To do so, the following studies have been under taken in this thesis:

- To actively control sound transmission through a soft-core sandwich panel excited by an acoustic excitation (plane wave), volume velocity cancellation control strategy is used, with the main focus on attenuating the flexural and dilatational modes and achieving sound attenuation in a broad frequency range.
- To actively control sound transmission through a soft-core sandwich panel excited by a structural excitation (line moment), two control strategies, minimization of volume velocity and WSSG at error sensor locations, are implemented and the results are compared. Reddy’s third order shear deformation theory is used to get a precise result, especially at dilatational modes which generally occur at high frequencies.
- Extensive study on volume velocity and WSSG is carried out in order to investigate how WSSG can achieve comparable amount of sound attenuation with volume velocity.
- To actively control sound transmission through a double panel partition, minimization of WSSG control strategy is used in order to attenuate the dipole-type motion in a double panel system. Also, acoustic point source arrangements are considered in the air cavity between the panels to further attenuate the transmitted sound power. The results are compared with the minimization of volume velocity.

1.3 Dissertation structure

This thesis first briefly overviews volume velocity, acoustic radiation modes and weighted sum of spatial gradient (WSSG) metrics. After this, a brief theory on soft-core sandwich panel is reviewed and the control strategies (volume velocity and WSSG) are applied on soft-core sandwich panel to control sound transmission through it (Publication I and II). How WSSG is able to accomplish a comparable amount of sound reduction with volume velocity is then explained (Publication III). At the end, a short concept of double panel partition and the effects of the implementation of the control strategies on sound transmission are presented (Publication IV). Finally, the thesis concludes with a brief discussion on the main results.

2. Active sound control theories

2.1 Overview of control metrics

2.1.1 Volume velocity

Focusing on the relationship between structural vibrations and acoustic radiation brings out two relatively well-known concepts. The first of these is the concept of volume velocity. Research has suggested that most of the acoustic radiation from a structure can be attributed to the more global quantity of volume velocity (Sors and Elliott, 2002; Johnson and Elliott, 1995; Elliott and Johnson, 1993; Guigou et al., 1996). This can be seen from Rayleigh's integral given as

$$p(\mathbf{r},t) = \frac{j\omega\rho_0}{2\pi} e^{j\omega t} \int_S \frac{\tilde{v}_n(\mathbf{r}_s) e^{-jkR_R}}{R_R} dS, \tag{2.1}$$

where p is the pressure, ω is the angular frequency in radians per second and ρ_0 is the density of the medium through which the sound is propagating. Also, $\mathbf{r}$ is the position vector of the observation point, $\mathbf{r}_s$ is the position on the surface, with a velocity amplitude $\tilde{v}_n$, and R_R is the magnitude of $\mathbf{r}-\mathbf{r}_s$. As can be seen, a reduction in the overall level of $\tilde{v}_n$ on the structure will tend to decrease the pressure at all points in the field. Volume velocity in its most basic sense refers to the net velocity of a vibrating structure. Thus, although in some instances, the amplitude of the vibration response may be greater, the volume velocity can be close to zero. Even modes will display this property while odd modes will not. As research has shown, odd modes radiate more efficiently than even modes because they have non-zero volume velocity. This is one of the reasons for volume velocity being strongly associated with acoustic radiation. Volume velocity of a flat plate-like structure can be determined by the summation of the product of velocity and the corresponding elemental area of each element. Therefore, the volume velocity of each element where the volume velocity sensors (e.g., accelerometers) are located can be written as (Johnson and Elliott, 1995; Sors and Elliott, 2002),

$$Q = a_x b_y \tilde{v}_n, \tag{2.2}$$

Where Q is the net complex volume velocity, $a_x = a/N_{sx}$ and $b_y = b/N_{sy}$ where N_{sx} and N_{sy} are the number of sensors placed in X and Y directions, respectively. So, the optimal secondary source strength to attenuate the radiated sound power can be found by minimizing the square of the volume velocity.

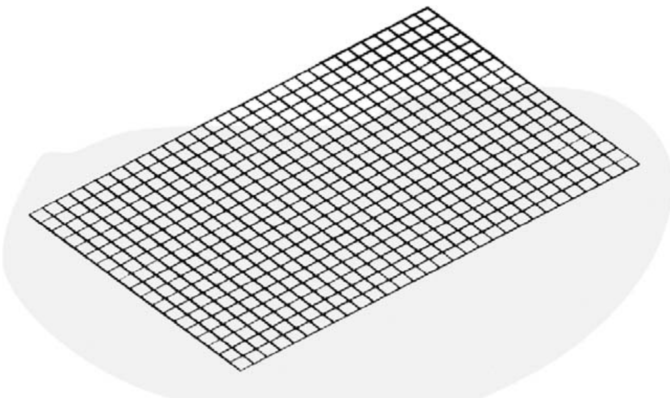

Figure 2.1. A panel divided into N piston elements.

2.1.2 Acoustic radiation modes

A second relationship between structural vibrations and acoustic radiation deals with acoustic radiation modes, which radiate sound power independently of the structural vibrations. Control of those modes gives a guarantee for the attenuation of sound power (Fahy and Gardonio, 2006). Elemental radiator formulation approach (Elliot and Johnson, 1993) is used to formulate the radiated sound power. In this formulation, a panel is divided into a grid of

N elements, as shown in Figure 2.1. The transverse vibrations of the elements are specified in terms of the velocities v_{en} at their center positions so that the overall vibration of the panel can be described by a column vector of complex amplitudes such as

$$\{\tilde{\mathbf{v}}_{\mathbf{e}}\} = [\tilde{v}_{e1}\ \tilde{v}_{e2}\ \dots\ \tilde{v}_{eN}]^T . \tag{2.3}$$

Therefore, the total radiated sound power can be defined as

$$\overline{\mathbf{P}}(\omega) = (\tilde{\mathbf{v}}_{\mathbf{e}})^H \mathbf{R}(\tilde{\mathbf{v}}_{\mathbf{e}}), \tag{2.4}$$

where matrix $\mathbf{R}$ is defined as the radiation resistance matrix, which discretizes the plate geometry and provides a simplified and computationally more efficient method for calculating the radiated sound power at low frequencies, and it is given by

$$\mathbf{R} = \frac{\omega^2 \rho_0 A_e^2}{4\pi c_0} \begin{bmatrix} 1 & \dfrac{\sin(kR_{12})}{kR_{12}} & \cdots & \dfrac{\sin(kR_{1N})}{kR_{1N}} \\ \dfrac{\sin(kR_{12})}{kR_{12}} & 1 & \cdots & \vdots \\ \vdots & \vdots & \ddots & \vdots \\ \dfrac{\sin(kR_{N1})}{kR_{N1}} & \cdots & \cdots & 1 \end{bmatrix} . \tag{2.5}$$

Since matrix $\mathbf{R}$ is real, symmetric, and positive-definite, the acoustic radiation modes can be obtained from the orthogonal decomposition of matrix $\mathbf{R}$ as

$$\mathbf{R} = \mathbf{Q}^T \mathbf{\Lambda} \mathbf{Q} \tag{2.6}$$

where $\mathbf{Q}$ is a matrix of orthogonal eigenvectors and $\mathbf{\Lambda}$ is a diagonal matrix of eigenvalues. The relative efficiencies of the radiation modes are given by the elements of $\mathbf{\Lambda}$, and the shape of each mode is given by the corresponding row of matrix $\mathbf{Q}$. The general shapes of the first six acoustic radiation modes of a rectangular panel are shown in Figure 2.2.

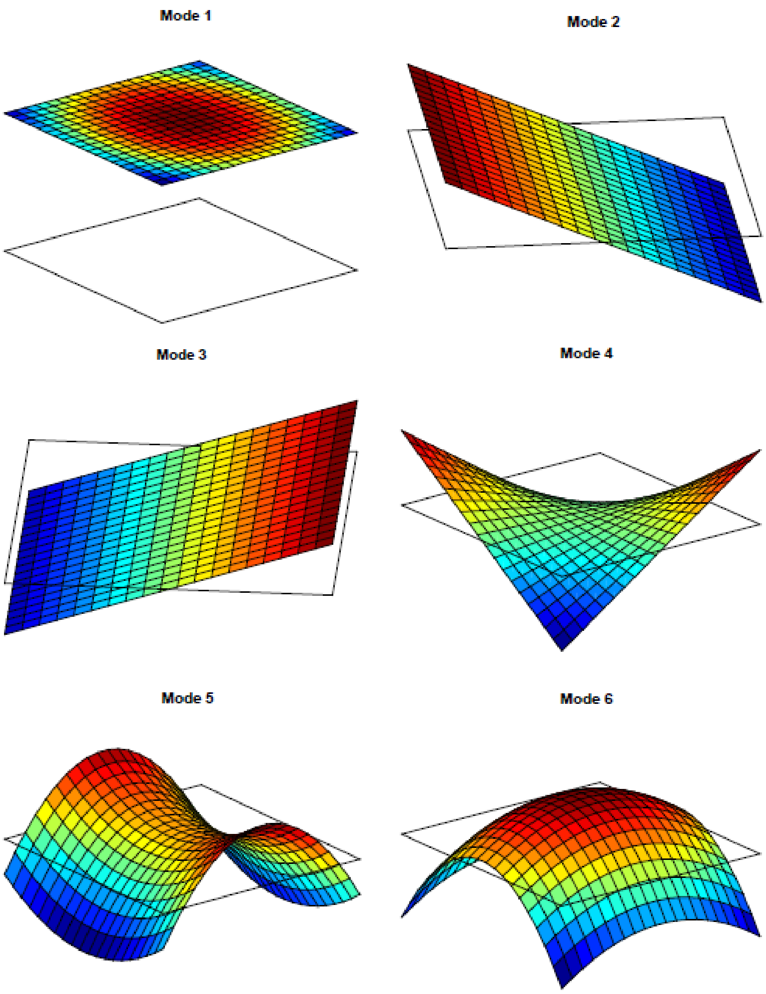

Figure 2.2. Acoustic radiation mode shapes.

Now Eq. (2.4) can be rewritten as

$$\bar{P}(\omega)=\left(\tilde{\mathbf{v}}_{\mathbf{e}}\right)^{H}\mathbf{Q}^{T}\mathbf{\Lambda}\mathbf{Q}\left(\tilde{\mathbf{v}}_{\mathbf{e}}\right)=\mathbf{y}^{H}\mathbf{R}\mathbf{y}=\sum_{r=1}^{D}\lambda_{r}\left|y_{r}\right|^{2}, \tag{2.7}$$

Where λ_r and y_r are the components corresponding to the radiation mode of interest. The shape of each radiation mode is mildly dependent on frequency. The higher the frequency, the more curvature appears in the individual radiation modes (Fahy and Gardonio, 2006). In order to compare the relative importance of the individual radiation modes, the power radiated by the individual acoustic radiation modes can be calculated as

$$\bar{P}_m(\omega) = \lambda_m |y_m|^2 , \tag{2.8}$$

with m being the index of the individual mode.

Controlling radiation modes has been an effective way to control the power radiated from a panel. However, the structural geometry associated with the vibrations must be known a priori to calculate the radiation modes and determine sensor locations that are conducive to sensing all significant radiation modes present. In most cases, structural vibrations cannot be fully mapped without equipment such as multiple accelerometer arrays or a scanning laser Doppler vibrometer, and the radiation modes cannot be obtained without some numerical analysis of the structure. Furthermore, depending on how many acoustic radiation modes are significant, these techniques can require the use of a large number of sensors which could be required to estimate the amplitudes of the significant radiation modes.

2.1.3 Weighted sum of spatial gradients (WSSG)

The analysis of both volume velocity and acoustic radiation modes supports the idea that a quantity which gives a vibration field related to volume velocity as well as mimics the acoustic radiation modes should be a desirable objective function to minimize. If this effect could be created using a point sensor measurement rather than a distributed array of sensors, a global result could be achieved using a simpler sensor configuration than for the other objective functions which, as previously mentioned, can require a large distributed array of sensors to estimate their respective quantities. A quantity that represents the volume velocity as well as mimics acoustic radiation modes has been developed, and called as weighted sum of gradient control metric (WSSG). WSSG of a structure is simply the sum of scaling values multiplied with the square of transverse velocity, rocking velocity in X and Y directions and twisting velocity, and it is given by (Fisher et al., 2012)

$$\mathrm{WSSG} = \alpha\left(\tilde{v}\right)^2 + \beta\left(\frac{\partial \tilde{v}}{\partial x}\right)^2 + \gamma\left(\frac{\partial \tilde{v}}{\partial y}\right)^2 + \delta\left(\frac{\partial^2 \tilde{v}}{\partial x \partial y}\right)^2, \tag{2.9}$$

where α, β, γ and δ are the scaling values. For a simply supported panel these quantities are defined as $\alpha = 1.0$, $\beta = (a / m_1 \pi)^2$, $\gamma = (b / m_2 \pi)^2$ and $\delta = (ab / m_1 m_2 \pi)^2$. The optimal secondary source strength to attenuate the radiated sound power can be found by minimizing the square of WSSG.

3. Active control of sound transmission through a soft-core sandwich panel

3.1 Theory of soft-core sandwich panel

Sandwich panels are made by bonding thin high-stiffness face plates to a low-stiffness but relatively thick core material. The main load-carrying portions are the skins, while the core serves as a spacer to keep the face plates at a sufficiently large distance from the neutral surface. As a result, the maximum normal stresses are in the skins and the core carries most of the transverse shear. The high stiffness-to-weight ratio makes this kind of structural element a very attractive design option in weight-critical structures such as those found in spacecraft and aerospace industries (Fahy and Gardonio, 2006). However, the main issue and challenge in noise control with thick-core sandwich panels is the control of shear waves that usually couple well with the acoustic waves, which leads to an increased radiation efficiency and lower sound transmission loss (mainly a consequence of a wide coincidence frequency zone). Additional sound absorption materials for these types of structures fail to provide satisfactory results in the low-frequency region, and therefore, active control techniques have been researched to increase the sound transmission loss of these kinds of structures (Petitjean et al., 2002).

Due to their flexibility in the manufacturing process and due to weight considerations, foams and non-metallic honeycombs replace some of the traditional metallic honeycomb cores. The major difference between a metallic honeycomb and a “soft” core is due to out-of-plane flexibility of the later that significantly affects the overall behavior, which under various loading schemes may lead to different behavior patterns in the upper and the lower face plates (Sokolinsky and Frostig, 1999).

A sandwich structure having two stiff thin skins with a soft, light, and shear-resistant core can be approximated as a mechanical system of two-degrees of freedom, a mass-spring-mass system, as shown in Figure 3.1.

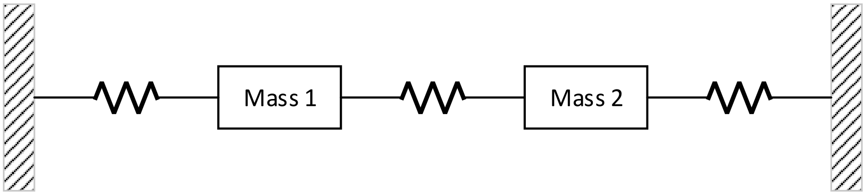

Figure 3.1. Mass-spring-mass equivalent system.

A system like the one in Figure 3.1 is characterized by two vibration modes: (i) one with both masses moving in the same direction, (ii) one with masses moving to the opposite directions. A sandwich structure behaves exactly in the same way. It is possible to identify two fundamental modes of vibration: (i) symmetric mode or dilatational mode due to core shear, (ii) anti-symmetric mode or flexural mode due to skin bending. For the sandwich construction with soft-core, the flexural vibration modes are governed by the stiffness of the two face plates and the out-of-phase dilatational modes are controlled by the stiffness characteristics of the core. Figures 3.2 and 3.3 show the flexural mode and dilatational mode of a soft-core sandwich panel, respectively. A general trend in case of vibration modes in a soft-core sandwich panel, which is investigated in this study, is that the flexural modes of vibration can be observed at lower natural frequencies (~1Hz to ~400 Hz) whereas the dilatational mode of vibration occurs at higher natural frequencies. The mode of vibration also plays a critical role in the sound transmission behavior of the sandwich structures. It has been noted that the soft-core sandwich panels exhibit noise transmission characteristics similar to those of double wall elastic panels except in the frequency range where dilatational mode occurs.

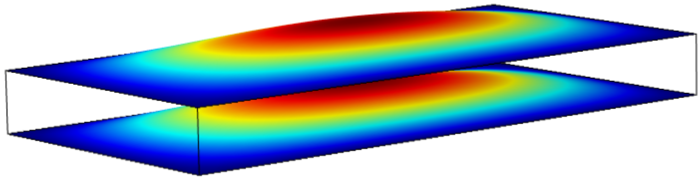

Figure 3.2. Flexural mode of a soft-core sandwich panel.

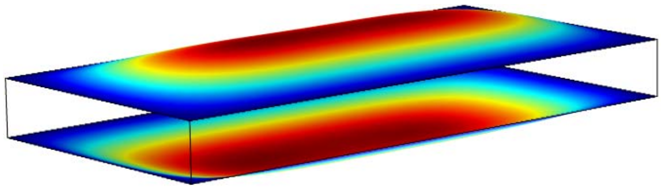

Figure 3.3. Dilatational mode of a soft-core sandwich panel.

In Publications I, II and III, a very soft viscoelastic core is sandwiched between two simply supported face plates. The term “soft-core” was introduced by Vaicaitis (1977) for cores whose Poisson’s ratio is nearly zero. Therefore, the motion of the core is only in the out-of-plane direction, and it acts merely as a viscoelastic spring. This has been deliberately done to excite the panel in both flexural and dilatational modes in order to attenuate sound in a large frequency band. Cellular honeycombs with zero Poisson’s ratio were proposed and realized in practice (Olympio and Gandhi, 2010). However, the control strategies used in this report can be implemented to any system (e.g., Publication IV) to attenuate the low-frequency sound transmission through it.

3.2 Control strategy implementation on a soft-core sandwich panel

As mentioned before, flexural modes occur in the low-frequency region and dilatational modes occur in the high-frequency region (Vaicaitis, 1977). The focus of this study is to control these modes and to achieve sound attenuation in a large frequency band. Classical plate theory (CPT) was used in Publication I, where it was found that flexural eigenfrequencies have a close agreement between finite element modeling (COMSOL Multiphysics) and analytical modeling, however, there is a difference in the dilatational eigenfrequencies.

It is well known that CPT does not take transverse shear deformations into account and thus yields artificially higher frequencies than the actual frequencies, especially at high frequencies where the effect of transverse shear deformation is significant. According to three-dimensional elasticity theory, transverse shear strains vary at least quadratically through the thickness of the plate. As a consequence, the shear correction factors were introduced to correct the discrepancy in the shear forces of the first-order shear deformation theory and the three-dimensional elasticity theory. But these shear correction factors depend on geometry, boundary conditions, and material properties, and they are very difficult to determine in complicated systems. The third order shear deformation theory developed by Reddy (2004), which accommodates quadratic variation of transverse shear strains and satisfies the boundary conditions so that the transverse shear stresses vanish on the top and bottom faces of a plate, eliminates the need for shear correction factors.

Therefore, Reddy's third order shear deformation theory is used in Publication II. Acoustic excitation (plane wave) is used as the primary force, point force and distributed force are used as the secondary force to cancel the volume velocity of the sandwich panel in order to attenuate sound pressure in a rectangular cavity which is attached to the sandwich panel (Publication I). On the other hand, in Publication II, a structural excitation (line moment which can be seen at various places, e.g., the air flow acting on aircraft wings may generate line moment at the fuselage) is used to excite the sandwich panel, and three piezoelectric actuators are used to feed the required control force to minimize the volume velocity and weighted sum of spatial gradients (WSSG). Since, the principle of sound control is the same in Publications I and II; therefore, brief results from Publication II are discussed here.

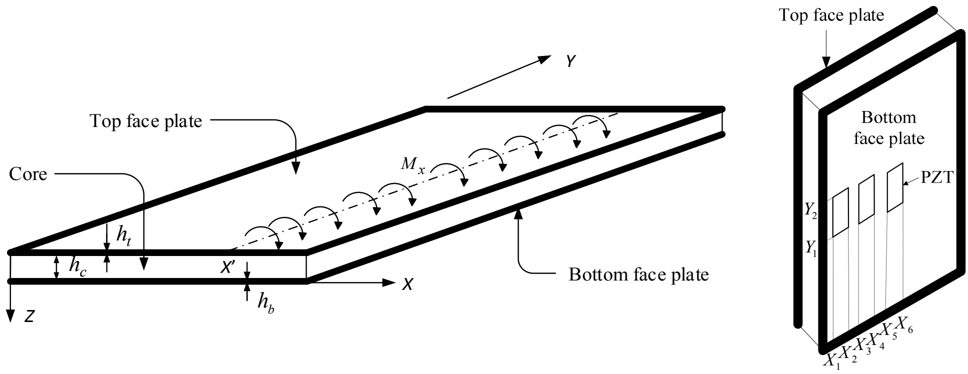

Figure 3.4. Sandwich panel excited by a distributed line moment and three PZTs.

Figure 3.4 shows a rectangular sandwich panel, which is made up of two thin face plates (i.e., the top and the bottom face plates) and a thick viscoelastic soft-core, where the top face plate is excited by a low-frequency distributed line moment of amplitude $1\ \mathrm{N\ m/m}$ and three PZTs are attached on the bottom face plate. The bottom face plate with the PZTs is considered as a laminated structure. Since a very soft viscoelastic core is considered, Poisson's ratio of the material is nearly zero; hence, the core can be approximated as a viscoelastic spring. In order to derive the governing equations of motion, the following assumptions are made:

1. The sandwich panel is flat and simply supported at all four edges.
2. The top and bottom face plates, viscoelastic core and piezoelectric patches are isotropic.
3. The neutral plane of the laminated structure coincides with the mid-plane of the bottom face plate.
4. The transverse normal strain of the laminated structure and top face plate is zero.
5. The transverse shearing strains of the laminated structure equal to zero at the top and bottom surfaces of the laminated structure. A similar theory holds for the top face plate as well.
6. Fluid loading is neglected.

7. Changes in the face plate thickness, if any, are neglected.

8. The sandwich panel is placed on an infinite rigid baffle.

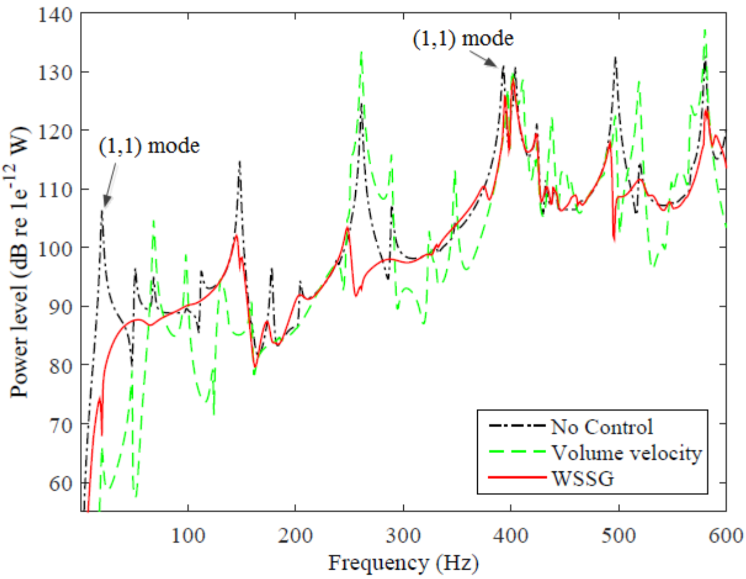

Figure 3.5. Radiated sound power when the top face plate is excited at $X' = 0.2$ m for $\xi = 0$. Key: (1,1) mode shown in the low frequency is the first flexural mode, (1,1) mode shown in the high frequency is the first dilatational mode.

The dimensions of the sandwich panel are the same as those by Vaicaitis (1977), where the sandwich panel is $0.25\ \mathrm{m} \times 0.508\ \mathrm{m}$ and the thicknesses of each face plate and the core are 0.51 mm and 6.35 mm, respectively. The material properties and the position of the PZTs can be found from Publication II. Volume velocity was determined by equally distributing 55 sensors, that is, 11 sensors in the *Y*-direction and 5 sensors in the *X*-direction. This number of sensors (e.g., accelerometers) is sufficient to accurately measure the volume velocity based on the methods described by Sors and Elliott (2002). One sensor is placed away from the corner of the plate, making sure that it won't overlap the PZT, to measure the WSSG, as it is done by Fisher et al. (2012). The sensor measures the desired parameter and passes the information to the control algorithm (WSSG as mentioned in Eq. 2.9). The control algorithm uses this information to calculate the proper signal for the PZTs. Now, the PZTs emit a new signal which drives the measurement at the sensor to a minimum. This process is repeated continuously. Average sound attenuation over the frequency range was calculated by integrating the total power over the frequency range with and without control. A total of 60 structural modes were used for the simulation in frequency range 0 to 600 Hz, as it is verified (see Sec. 7.1 in Publication II). In Figures 3.5, 3.6 and 3.7, X' and ξ represent the location of the line moment and isotropic core loss factor.

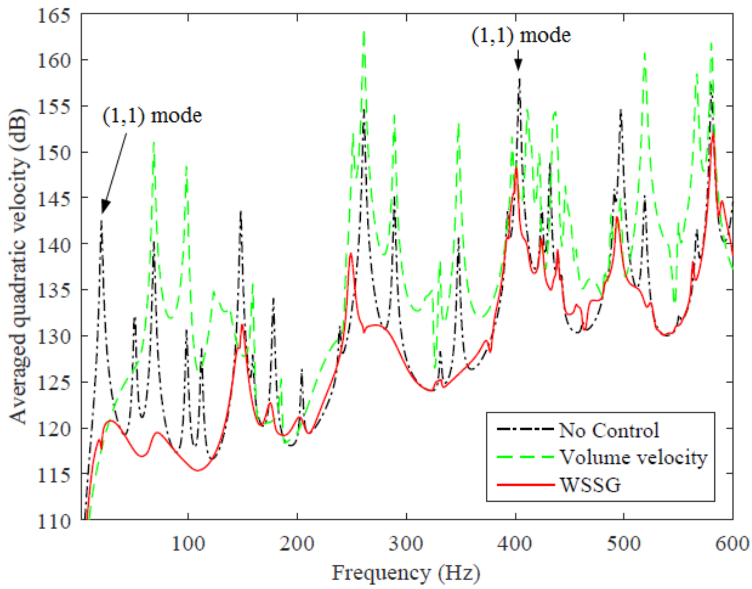

Figure 3.6. Averaged quadratic velocity of the bottom face plate when the top face plate is excited at $X' = 0.2$ m for $\xi = 0$. Key: as for Figure 3.5.

Figure 3.5 represents the simulated radiated sound power level from the bottom face plate at $\xi = 0$ with and without the minimization of volume velocity and WSSG at the sensor locations. For elastic sandwich panel $(\xi = 0)$ at first flexural and dilatational mode, the radiated sound power in WSSG is 67 dB and 125 dB, respectively, the corresponding values for volume velocity being 66 dB and 122 dB. While the actuators are driven to cancel the small amount of volume velocity at frequencies of 68 Hz and 261 Hz, the PZTs strongly resonate the panel in an excitation with (2,1) and (4,1) modes, respectively. Therefore, the average sound attenuation in WSSG is 4.06 dB, whereas for volume velocity there is an increase in 1.76 dB of average sound attenuation occurs. By looking at these numbers, it can be concluded that, at first flexural and dilatational mode, volume velocity strategy works better than WSSG. However, at other natural frequencies, WSSG outperforms volume velocity because WSSG targets a multiple number of modes as opposed to just the first radiation mode, which is what the volume velocity does. Also the maximum increase in radiated power is less in WSSG, which is an important consideration for cases where the structural excitation is narrow-band in nature. Similar type of analysis was carried out by varying the core loss factor to see the effect of the control strategies on viscoelastic sandwich panel (see Publication II for more details).

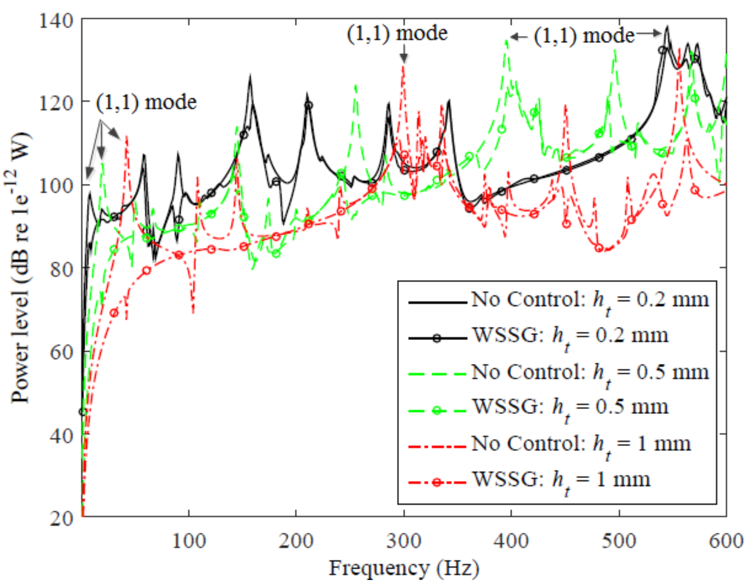

Figure 3.7. Effect of panel thickness on radiated sound power when the top face plate is excited at $X' = 0.2$ m for $\xi = 0$. Key: as for Figure 3.5.

To explore the effectiveness of the control strategies on the vibration velocity of the radiating face plate, averaged quadratic velocity of the same was calculated, and it is shown in Figure 3.6. This figure shows the averaged quadratic velocity of the bottom face plate as a function of frequency when the top face plate is excited by a line moment along $X' = 0.2$ m for zero core loss factor $(\xi = 0)$. At $(1,1)$ structural mode, the vibration velocity of the bottom face plate is a little bit less after the minimization of volume velocity as compared to the minimization of WSSG, which verifies why the radiated sound power is less at 19 Hz in Figure 3.5. However, after that, WSSG works extremely well to attenuate the vibration velocity of the bottom face plate and hence, it is able to mitigate the radiated sound power in the whole frequency range considered here (see Figure 3.5).

Also, the effect of face plate thickness on the radiated sound power is explored. By selecting the thickness of the top and bottom face plates for three values, 0.2 mm, 0.5 mm and 1 mm, and keeping the other geometrical dimensions of the structure fixed, the radiated sound power level as a function of frequency is plotted when the line moment is excited along $X' = 0.2$ m on an elastic sandwich panel $(\xi = 0)$. This is shown in Figure 3.7. As it has already been concluded that WSSG works better than volume velocity, only WSSG control method is considered in this investigation. It can be seen from the figure that the frequency gap between the first flexural and dilatational modes is reduced with the increase in face plate thickness. At the first flexural mode, the radiated sound power before control is highest for the thicker face plate among the three configurations considered here; however, it has the lowest radiated sound power at the first dilatational mode. With the increase in thickness, the controlled radiated sound power decreases, and hence the sound attenuation over the frequency range is 11.7 dB, 6.1 dB and -0.6 dB for face plate thickness of 1 mm, 0.5 mm and 0.2 mm, respectively.

Table 3.1. Scaling factors and average sound attenuation for WSSG at $\xi = 0$.

	α	β	γ	δ	Avg. sound attenuation in dB (1- 450 Hz)
Avg. value (set-1)	1.0	0.00219	0.00445	8.36×10^{-6}	4.55
Avg. value (set-2)	1.0	0.00428	0.0105	3.81×10^{-5}	4.77
Avg. value (set-3)	1.0	0.00446	0.0261	1.16×10^{-4}	7.25
Avg. value (set-4)	1.0	0.00633	0.0261	1.65×10^{-4}	-8.01

3.3 WSSG to achieve comparable result with volume velocity

In Sec. 3.2, it was seen that the WSSG control metric has the ability to control multiple acoustic radiation modes and thus is able to attenuate sound convincingly. However, in some instances (e.g., when the panel excites in (odd, odd) modes only), the volume velocity control strategy attenuates more sound than WSSG. Since it has already been verified that the control strategy of minimizing the volume velocity is difficult to implement in practice (Fisher et al., 2012), this section will investigate how WSSG can achieve comparable sound attenuation with volume velocity.

An oblique plane wave of pressure amplitude of 1 Pa is incident on the top face plate of a soft-core sandwich panel. It can be seen from the figures below that most of the sound is radiated at (odd, odd) modes; therefore, a thin piezoceramic actuator (PZT) is attached exactly at the center of the bottom face plate. The optimal magnitude and phase of the necessary voltage supplied to the PZT to minimize the volume velocity and the WSSG at the sensor locations have been analytically calculated. In addition, the constraint condition for the amplitude of voltage supplied has been limited to 100 Volts. All the results are presented graphically in two ways: (i) radiated sound power versus frequencies before and after control and (ii) voltage required to minimize the volume velocity and WSSG versus frequencies.

Also, the average sound attenuation over the frequency range is calculated by integrating the total sound power over the frequency range with and without control. In the figures below, α, β, γ, and δ represent the scaling factors (as defined in Sec. 2.1.3) used in the expression for WSSG.

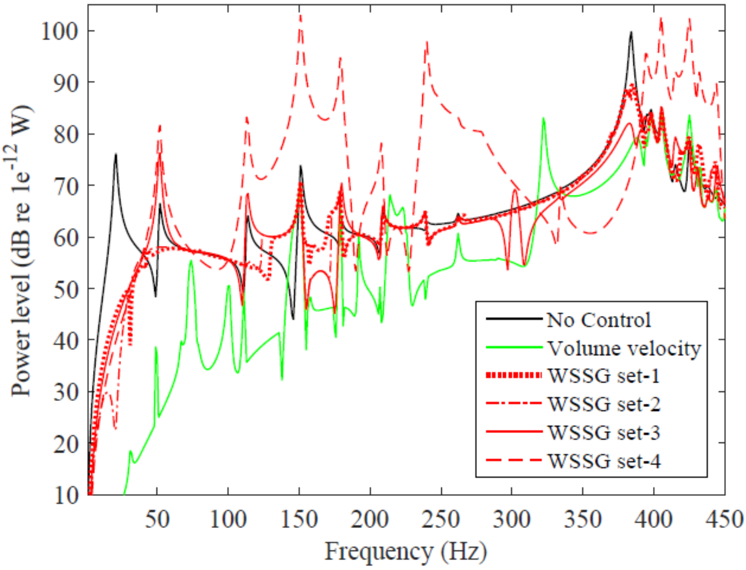

Figure 3.8. Simulated radiated power using WSSG and volume velocity for $\xi = 0$.

Radiated sound power level from elastic soft-core sandwich panel $(\xi = 0)$ with and without the control has been analytically calculated, and it is shown in Figure 3.8. In this figure, from the black solid line, which shows the radiated sound power before the implementation of any control metric, it can be observed that the sound is basically radiated from the bottom face plate at (odd, odd) modes: (1,1), (1,3), (1,5) (3,1), (3,3) (1,7), (3,5), and (1,9) are for flexural modes and (1,1), (1,3) and (3,1) are for dilatational modes. Out of these (1,1) and (3,1) flexural modes and (1,1) dilatational mode have very high peaks, i.e., these modes have more contributions to average sound power. Therefore, this study considers 4 sets of modes to examine the effect of different scaling factors on sound power attenuation.

In set-1, 45 modes are considered as was done by Fisher et al. (2012). In set-2, all the peaks are taken: (1,1), (1,3), (1,5) (3,1), (3,3) (1,7), (3,5), and (1,9) are for flexural modes and (1,1), (1,3) and (3,1) are for dilatational modes. In set-3, (1,1) and (3,1) flexural modes and (1,1) dilatational mode are considered. And in set-4, to see the effect on first flexural and dilatational modes where volume velocity control method works significantly well from the sound attenuation point of view, only (1,1) flexural mode and (1,1) dilatational mode are considered. The scaling factors and the average sound attenuation for all the sets have been calculated, and are shown in Table 3.1. We consider 45 structural modes to obtain the objective function for volume velocity, and the average sound attenuation is calculated as 8.81 dB.

In Figure 3.8, the dotted and the dash-dot lines are almost overlapping each other, which shows that instead of taking all the modes in the frequency range like in the volume velocity case, scaling factors calculated using the indices of all the modal peaks would yield similar result. Both these sets of modes are able to control the flexural modal peaks and also some low-order dilatational peaks; however, the average sound attenuation in WSSG is approximately 51% of the volume velocity control because both sets weigh all the modes equally but do not give importance to the high-level peaks. For example, set-1 and set-2 attenuate only 10 dB of sound at the first dilatational mode, whereas in set-3, which only considers the most sound radiating peaks, the WSSG control metric attenuates around 18 dB of sound power although it increases a little bit at (1,3) and (1,5) flexural modes which bring about an increase of the average sound attenuation to 82% of the volume velocity. By comparing the solid green line and red dashed line, which represent the volume velocity and the WSSG for set-4, it can be observed that at the first dilatational mode, the radiated sound power for WSSG is 72 dB whereas for volume velocity it is 77 dB. However, the average sound attenuation is decreased in WSSG since there is a significant increase in sound at other modes. Therefore, it can be concluded that to achieve maximum sound attenuation, modes should be intelligently selected while calculating the scaling factors in WSSG, as explained in Publication III.

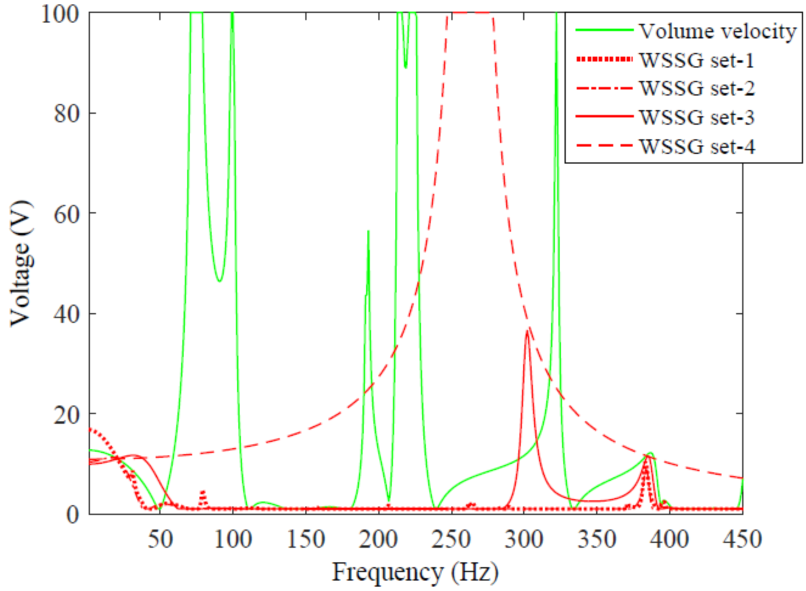

Figure 3.9. Voltage required to minimize WSSG and volume velocity for $\xi = 0$.

Figure 3.9 depicts the required voltage to be fed to the actuator to minimize the volume velocity and WSSG at sensor locations. The maximum necessary voltage for volume velocity control is 100 Volts, whereas maximum 35 Volts are needed in WSSG for set-3, which represents the set of modes that attenuates the sound power close to the volume velocity. This is an important consideration for the PZT since it is known to exhibit non-linearity at high voltage. However, for set-4, the maximum voltage required is 100 Volts, which is due to the large change in radiated sound power over the frequency range. A similar type of analysis was carried out by varying the core loss factor (see Publication III for more details).

4. Active control of sound transmission through a double panel partition

4.1 Theory on double panel partition

The transmission of sound energy in a separation element proceeds by the vibration of the element, with the mass and sound frequency being relevant variables. As the mass of the element increases, so does insulation, as a result inertial force increases. When the frequency of sound incident on an element that maintains the same mass is increased, the vibration power of the element decreases and greater dissipation of sound energy is observed, leading to the rise in acoustic insulation (Tadeu and Mateus, 2001).

Besides these two variables, there are others that may affect the acoustic insulation of a separation element. These include the angle of incidence of the waves, the existence of weak points in the insulation, rigidity, damping of the element; and in the case of multiple elements, the number of panels and their individual characteristics and separation. Vibro-acoustic problems involving the interaction of structural vibration and acoustic-structural coupling effects are of particular significance in engineering applications. A typical problem of this kind concerns a configuration comprising two parallel rectangular panels with an airproof cavity sealed in between, mounted on acoustic rigid baffle, with the whole system immersed in a fluid (e.g., air). These double panel partitions have found increasingly wide applications in modern buildings, vehicles, aerospace and aeronautical structures, etc., due to their superior sound insulation properties over single panel configurations, as a result of structural discontinuity and sound impedance mismatch. However, there remains a problem with the double panel: it has weak sound transmission loss (STL) performance at low frequency, due to the “mass-air-mass” resonance. This causes the double panel to lose its superiority over the single panel. Sound absorption materials in the air gap fail to provide satisfactory results in the low-frequency region because acoustic wavelengths are much longer than the dimension of damping materials. To get rid of this problem, active control techniques which increase the sound transmission loss of these kinds of structures have been investigated (Gardnnio and Elliot, 1999; Carneal and Fuller, 2004).

4.2 Control strategy implementation on double panel partition

To control the transmitted sound power through a double panel partition, cancellation of volume velocity of the radiating panel was proposed by Pan et al. (1998). They found that the double-leaf construction provides good passive attenuation of the first radiation mode at high frequencies; however, it couldn’t do much to control the even modes which make a dominant contribution to the radiated sound power and thus, there is no advantage in controlling volume velocity at high frequencies. Hence, they concluded that to attenuate sound in a large frequency band, two aspects need to be considered. One is the active structural acoustic control of the second and third radiation modes, which the volume velocity control strategy couldn’t detect. And the other one is to introduce acoustic sources in the air gap between the panels to control the (1,0,0) and (0,1,0) acoustic modes. In Publication IV, work done by Pan et al. (1998) was extended to accommodate these two facets of the problem. A WSSG control metric is used to control multiple acoustic radiation modes so that the dipole-type of motion of double panel systems which generally occurs at high frequencies can be attenuated. In addition to this, acoustic sources are introduced in the cavity to control the cavity modes. A piezoceramic actuator (PZT) is placed on one side of the panel to minimize the volume velocity and WSSG of the panel surface at error sensor locations. Also, effects of loudspeaker arrangements are discussed. And in the end, the effect of air cavity thickness and the angle of incidence of the incident sound wave on the controlled radiated sound power were investigated.

In order to demonstrate the effectiveness of WSSG, it has been compared with volume velocity, the most commonly used control metric for ASAC. The optimal magnitude and phase of the voltage supplied to the PZT is determined using a simple-gradient based algorithm for minimizing the square of WSSG and volume velocity at the error sensor locations. Volume velocity is calculated by equally distributing 30 sensors (e.g., accelerometers), that is, 6 sensors in the *X*-direction and 5 sensors in the *Y*-direction. This number of sensors is sufficient to give an accurate measure of volume velocity (Sors and Elliott, 2002). One sensor is placed away from the corner of the plate, making sure that it won't overlap the PZT, to measure the WSSG, as it is done by Fisher et al. (2012). For further comparison of results, the sound transmission loss is calculated by integrating the power transmission ratio over the frequency range and taking the reciprocal of the same before and after the implementation of the control strategy.

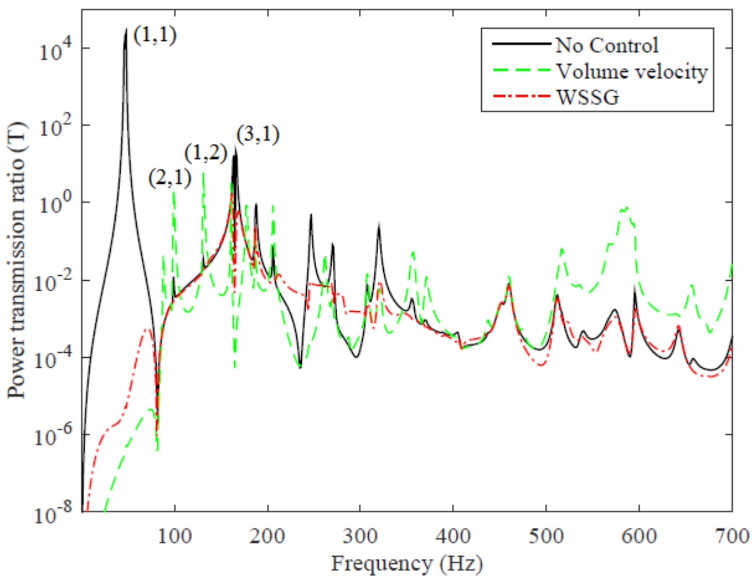

Figure 4.1. Sound power transmission ratio by driving a PZT on panel 2.

It has been verified that minimization of the control strategies on the radiating panel (say as panel 2) yields slightly better sound attenuation than minimizing the same on the incident panel (say as panel 1), see Publication IV for more details. The simulations also showed that positioning the PZT on the incident panel achieves the best results. Based on these two observations, the number of parameters can be reduced, and in the analysis below the minimization of control strategies (volume velocity or WSSG) is carried out on panel 2 only, and the PZT is placed on panel 1 only.

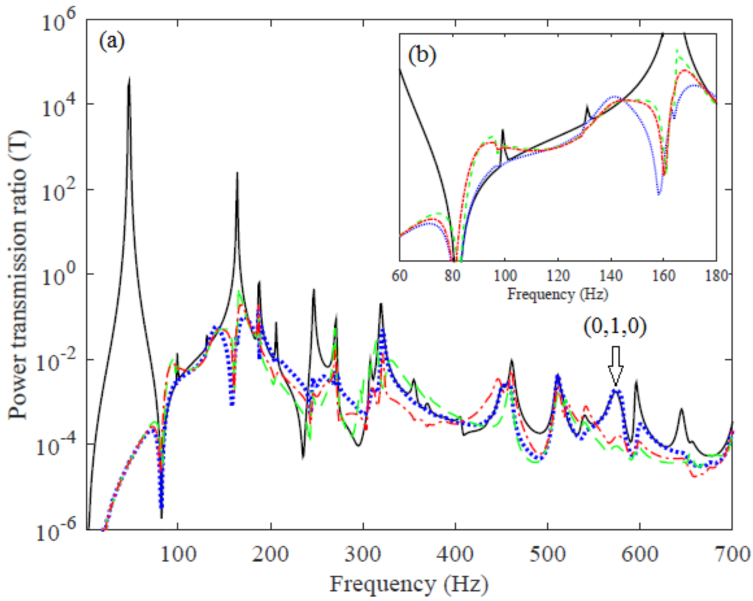

Figure 4.2. Sound power transmission ratio by driving a PZT on panel: (a) Full view (b) Zoom view from 60 to 180 Hz. ——— No control; ▪▪▪▪▪▪ structure controlled; — — — structural-cavity controlled with one loudspeaker; -·-·- structural-cavity controlled with three synchronized loudspeakers.

Minimization of volume velocity of the radiating panel at the sensor locations attenuates the radiated sound power especially in the low-frequency region; however, it is unable to perform like the WSSG in the whole frequency range considered here, which can be seen from Figure 4.1. For example, the volume velocity control method works better than the WSSG at (1,1) and (3,1) modes. However, at other frequencies, WSSG outperforms volume velocity because WSSG is able to control multiple acoustic radiation modes while volume velocity targets the first acoustic radiation mode only. This has been verified by the radiation mode analysis (Publication IV). At (2,1) and (1,2) panel modes (acoustic potential energy level is very low; see Publication IV), the actuator is driven to cancel the small amount of volume velocity of the radiating panel and, therefore, it strongly resonates the higher order modes of panel 2. Thus, the sound transmission loss in WSSG is 42 dB whereas for volume velocity it is 37 dB.

To further enhance the sound transmission performance, both structural and cavity control methods are simultaneously applied to the system. This is referred to as structural-cavity control. To analyze the results, power transmission ratio for the structural control and structural-cavity control are compared as shown in Figure 4.2. Since WSSG works better than volume velocity control strategy, only the WSSG control method is considered in the investigation below. Two types of (acoustic monopole point source) control loudspeaker arrangements were researched: in one of them, a loudspeaker is placed at $(0.3a, 0.3b, 0.3d)$ and, in the other, a uniform control pressure field is created by placing three synchronized loudspeakers (that is, three monopole acoustic point sources) triangular-symmetrically at $(0.3a, 0.3b, 0.3d)$, $(0.5a, 0.7b, 0.3d)$ and $(0.7a, 0.3b, 0.3d)$, where a and b are the length and breadth of each panel, respectively and d is the distance between the panels. Controlled by the PZT alone (structure controlled), it was possible to mitigate the power transmission ratio at most of the frequencies considered in this study; however, it wasn't possible to alter the sound power at 571 Hz, i.e., at (0,1,0) acoustic mode. On the other hand, both loudspeaker arrangements quite convincingly suppressed this mode and reduced the sound power transmission from 550 Hz to 700 Hz. One important observation can be made by looking closely at the figure in the low-frequency region (i.e., in between 0 to 180 Hz) shown in Figure 4.2(b). Both loudspeaker arrangements were able to decrease the sound power transmission in the whole frequency range. However, they excite the (0,0,0) acoustic mode, which is why the structural-cavity control transmits more sound power when compared to the structural control in the low-frequency region. Nevertheless, in the low-frequency region (Figure 12(b)), i.e., between 0 to 180 Hz, the symmetric arrangement of loudspeakers achieves a better control effect than the single loudspeaker, which is in good agreement with Li and Cheng (2008).

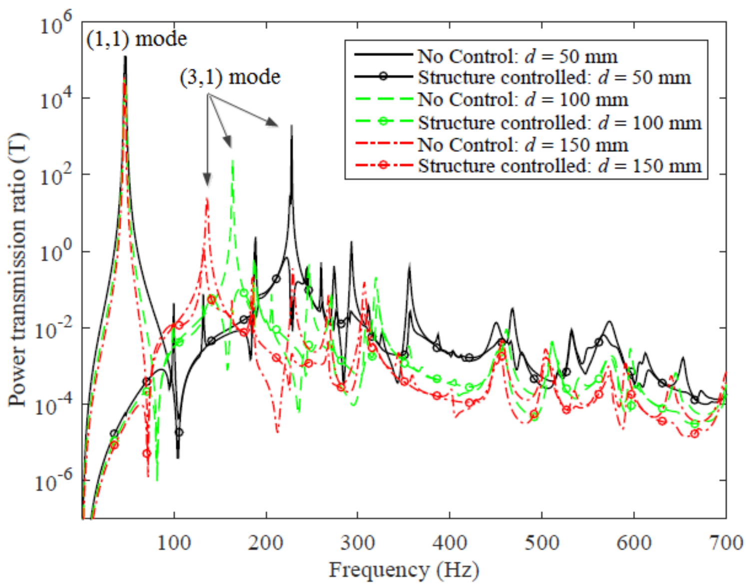

Figure 4.3. Effect of cavity thickness on sound power transmission ratio by driving a PZT on panel 1.

Transmission of sound through a double panel partition without any mechanical connection is due to the enclosed air between the panels. Air in the cavity acts as springs, and thus, transmits the mechanical vibration from the incident panel to the radiating panel and vice-versa. The equivalent stiffness of the air between two parallel panels is given by

$K_a = \rho_a c_a^2 / d$, where ρ_a and c_a are the density and speed of sound in air, respectively; hence, the equivalent stiffness of the air is expected to have a significant effect on the sound transmission. With other geometrical dimensions of the structure apart from the cavity thickness for three selected values, i.e., $d = 50$ mm, 100 mm and 150 mm being fixed, the power transmission ratio of the double panel partition as a function of frequency for the structural control (control by the PZT only) can be plotted, and shown in Figure 4.3.

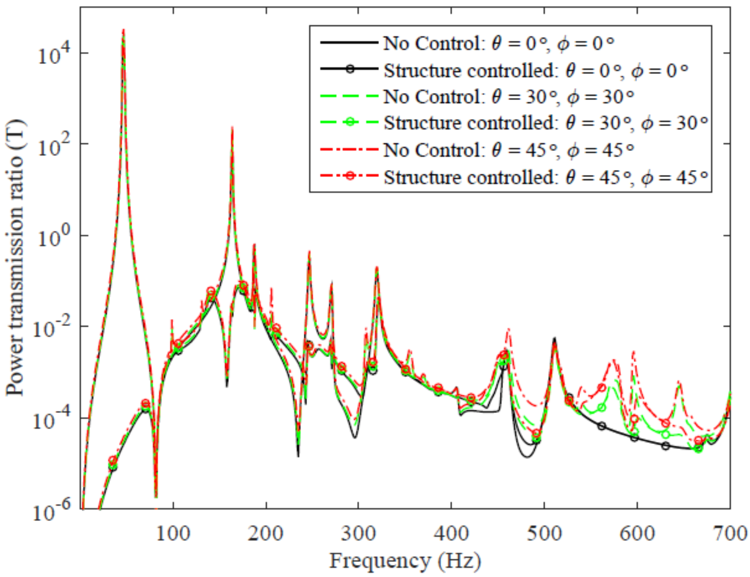

Figure 4.4. Effect of elevation angle on sound power transmission ratio by driving a PZT on panel 1.

It can be observed from Figure 4.3 that the first peak before control is unchanged by the variation in the thickness of the air cavity, which clearly shows that the vibro-acoustic coupling effect on this mode is very negligible. However, the (3,1) modal peak before control has been shifted towards left, with increase in the cavity thickness. The vibro-acoustic field inside the cavity seems to play a dominant role at (3,1) mode, and, hence, increase in the air cavity thickness leads to reduction in air stiffness, which causes a decrease in resonance frequency. Also, as expected, the uncontrolled power transmission ratio at (3,1) mode decreases with increase in the air cavity thickness. It can also be seen that when the thickness of air cavity goes on increasing, which further weakens the effect of vibro-acoustic coupling, it leads to further reduction in the controlled sound power transmission ratio.

In the examples discussed so far, only a plane wave incident at $\theta = \phi = 45^0$ has been considered. In order to explore the effect of the direction of plane wave on sound power transmission ratio, three selected values of equal elevation and azimuthal angle were considered; i.e., $\theta = \phi = 0^0$, 30^0 and 45^0. Structural control (PZT driven by WSSG control strategy) was implemented on the double panel system, and sound power transmission ratio as a function frequency is plotted in Figure 4.4. It can be noticed that, with the simultaneous increase in azimuthal and elevation angle, the sound power transmission goes on increasing especially at high frequencies. The plane wave normal to panel 1 $\left(\theta = \phi = 0^0\right)$ is unable to excite the (0,1,0) cavity mode, and hence, the solid lines overlap each other from 500 Hz to 700 Hz. Since WSSG was able to attenuate multiple acoustic radiation modes but not all the modes, the best result is achieved when the incident sound is normal to the panel plane because, at this angle of sound wave, the panel only vibrates with (odd, odd) modes and can be controlled by the minimization of volume velocity only (Pan et al., 1998).

5. Conclusions

In this thesis, numerical studies on active control of low-frequency sound transmission through soft-core sandwich panel and double panel partition were carried out. The use of conventional noise control technologies (acoustic insulation, constrained layer damping, etc.) in the lower frequency range is limited. An alternative noise control solution, requiring less additional weight and volume, is provided with the concept of active structural acoustic control (ASAC). These systems are based on the principle that structural sound radiation can be reduced by controlling the vibration response of a structure with appropriate actuators and sensors. Subsequently, a detailed study on two commonly used control metrics in ASAC, namely, volume velocity and weighted sum of spatial gradients (WSSG), which are intended to increase the low-frequency transmission loss, has been described in this thesis. The main original findings and features presented in this thesis are the following:

1. When coupled and uncoupled eigenfrequencies of a soft-core sandwich panel were compared, it was found that dilatational coupled eigenfrequencies are smaller than dilatational uncoupled eigenfrequencies. However, for flexural modes, this trend is reversed (Publication I).
2. Volume velocity cancellation control metric was able to control both the flexural and dilatational modes, and hence able to attenuate sound pressure in a wide frequency band (Publication I).
3. While the sandwich panel is excited by a line moment at the vertical midline of the top face plate, volume velocity control strategy is unable to reduce the radiated sound power

irrespective of core loss factors. However, WSSG control metric is able to attenuate the flexural and dilatational modal peaks and hence, reduce the radiated sound power over a large frequency band (Publication II).

4. On the other hand, while the line moment is acting off the midline, both simulated control metrics are able to control the modal peaks to attenuate the sound. However, WSSG achieved improved control over volume velocity at natural frequencies and modes higher than the fourth mode (Publication II).
5. By making "appropriate" choices of modes for the calculation of scaling factors, WSSG can provide comparable control to volume velocity. However, the necessary voltage required for the actuator to minimize the WSSG is less than the voltage required to minimize the volume velocity (Publication III).
6. Sound radiation mode analysis verifies that WSSG is able to attenuate dipole-type motion of the radiating panel by targeting multiple sound radiation modes, and, hence, provide better sound power transmission loss when compared to volume velocity control strategy. Addition of the acoustic sources inside the cavity is able to control the cavity modes and hence, increase sound transmission loss in a wider frequency band (Publication IV).
7. Multi-control acoustic monopole sources driven by a synchronised single-channel control yields a better control result at low freqencies when compared to a single acoustic point source (Publication IV).
8. Air cavity thickness strongly affects the overall vibro-acoustic behavior of the double panel system. The increase in air cavity thickness, which results in weakening vibro-acoustic coupling, leads to further reduction in the controlled sound power transmission ratio (Publication IV).

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